Design Method for the Heating/Cooling Coil in the AHU Based on Fuzzy Logic —Part Two: Design of the Minimum Heat-Exchanging Unit[1]

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Abstract: Considering a heating/cooling coil with adjustable heat-exchange area, an unequal type is put forward in this paper. Aiming at the application of such heat exchanger in an air-handling unit, restriction conditions are given for the minimum heat-exchanging unit in accordance with the requirement of control precision of indoor temperature and humidity. The structure adjustable heat exchanger improved the hydraulic adjusting characteristics of existing air handling units in the respective structure, and overcame the problems of existing air-handling units such as a narrow hydraulic adjusting range due to the rapid-opening performance of the continuous motor-driven value. As a result, such kind of heat exchanger is extremely suitable to fuzzy control.

Key words: coil; adjustable structure; minimum unit.

1. INTRODUCTION

The paper[1] advances a heating/cooling coil with unequal adjustable area as shown in Fig.1, where 1 is on/off control valve, 2 first manifold, 3 second manifold, $\overline{F}$ denotes fundamental heat-exchanging area (m^2), $\widetilde{F}_{\mathrm{S}}$, $\widetilde{F}_{\mathrm{M}}$ and $\widetilde{F}_{\mathrm{B}}$ denote adjustable heat-exchanging area (m^2), and $\widetilde{F}_{\mathrm{S}}$ means "small" heat-exchanging area (m^2), it is minimum heat-exchanging unit, $\widetilde{F}_{\mathrm{M}}$ means "medium" heat-exchanging area (m^2), and $\widetilde{F}_{\mathrm{B}}$ "big" heat-exchanging area (m^2). $\widetilde{F}_{\mathrm{S}}$, $\widetilde{F}_{\mathrm{M}}$ and $\widetilde{F}_{\mathrm{B}}$ constitute adjustable heat-exchanging area $\widetilde{F}$, $\overline{F}$ and $\widetilde{F}$ constitute the total heat-exchanging area F, namely

$$F = \overline{F} + \widetilde{F} \tag{1}$$

$$\begin{cases} \widetilde{F} = \widetilde{F}_S + 2\widetilde{F}_M + \widetilde{F}_L \\ \widetilde{F}_M = 2\widetilde{F}_S \\ \widetilde{F}_L = 2\widetilde{F}_M + \widetilde{F}_S \end{cases} \tag{2}$$

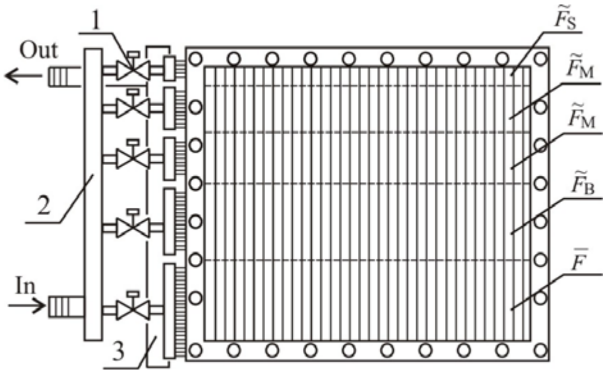

Fig. 1 The unequal adjustable area of the coil

According to equation (2), $\widetilde{F}$ would be calculated as soon as minimum heat exchanging unit $\widetilde{F}_{\mathrm{S}}$ can be got.

Adjustment of the coil structure will impact on both the supply air temperature of AHU and its humidity. Only $\widetilde{F}_S$ meets the requirements that temperature and humidity variations indoor caused by $\widetilde{F}_S$ changes are less than their control precision, can

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we satisfy the demands of control precision of temperature and humidity indoor while energy is regulated in AHU. The restriction conditions for $\widetilde{F}_S$ is given in Equation (3), where Δt_r denotes temperature variation indoor brought about by controlling the minimum heat-exchanging unit, that is temperature deviation (°C), t_r denotes temperature indoor (°C), t_{rg} denotes set point value of temperature indoor (°C), δ_t denotes control precision for temperature indoor (°C), $\Delta\varphi_r$ denotes relative humidity variation indoor raised by controlling the minimum heat-exchanging unit, that is relative humidity deviation (%), φ_r denotes relative humidity (%), φ_{rg} denotes set value of relative humidity indoor (%), δ_φ denotes control precision for relative humidity indoor (%).

$$\begin{cases}\Delta t_r = t_r - t_{rg} \le \delta_t \\ \Delta\varphi_r = \varphi_r - \varphi_{rg} \le \delta_\varphi\end{cases} \quad (3)$$

The variations of temperature and humidity are relevant to the factors such as dynamic load indoor, supply air parameters, exhaust air condition, and etc. while supply air parameters are defined by fresh air fraction, heat-transfer content of air-conditioning units and humidifying quantity (winter), etc. Hence we should analyze dynamic characteristics of the whole air-condition system to make the minimum heat-exchanging unit satisfy Equation (3).

2. MATHEMATICAL MODE FOR SUPPLY AIR TEMPERATURE

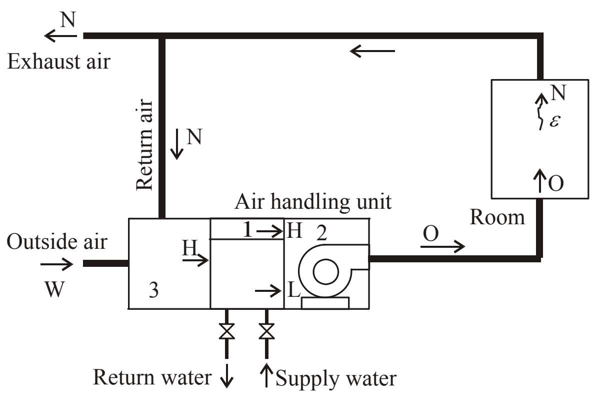

Fig.2 Scheme of an air-conditioning system with primary return air

Fig.2 shows an air-conditioning system with primary return air, where N/W denotes the air condition indoor/outdoor, H denotes mixed air state, L denotes apparatus dew point, O denotes supply air state, ε denotes ratio between cool load and humidity load indoor, 1 denotes the minimum heat-exchanging unit $\widetilde{F}_S$, 2 denotes fan mixed section, 3 denotes mixed air section.

For convenience of calculation, it is supposed each component in air-conditioning system (fan, coil etc) is able to meet the demands of air-conditioning design condition, energy loss in air-duct is small enough to be neglected; the air distribution and leakage indoor is ignored; face air velocity at different section of air-conditioning units are equivalent; heat transfer of heat-exchanging unit is zero while its valve is closed, and other unit will not be influenced.

Consider an air-conditioned room, we can get the following relation from the first law of thermodynamics:

The change of sensible heat indoor = the sensible heat input+ the sensible heat produced by room - the sensible heat output (4)

2.1 The Change of Sensible Heat Indoor

When temperature indoor $t_r(\tau)$ changes, it will cause the change of sensible heat indoor, as is given by Equation (5) (noted in difference scheme), where Q_r denotes sensible heat amount indoor (KW); τ denotes time variable (s); ρ_r denotes air density indoor (Kg/m^3); C_P denotes air specific heat at constant pressure (KJ/Kg·°C); V_r denotes air-conditioned room volume (m^3); T denotes sampling period (s).

$$\frac{dQ_r(\tau)}{d\tau} = \rho_r C_P V_r \frac{t_r(\tau) - t_r(\tau-1)}{T} \quad (5)$$

2.2 The Sensible Heat Input

The sensible heat input is mostly brought by supply air, as is given by Equation (6), where Q_O denotes the sensible heat amount carried by supply air (KW); ρ_O denotes supply air density (Kg/m^3), G denotes supply air volume (m^3/s); t_O denotes supply air temperature (°C).

$$Q_O(\tau) = \rho_O C_P G t_O(\tau) \tag{6}$$

2.3 The Sensible Heat Produced in Room

There are two types of sensible heat indoor, one is instantaneous cooling load carried by heat transfer, which has been known[3], noted as $Q_L(\tau)$ (KW); the other is instantaneous storage cooling load carried by storage input, noted as $Q_{SL}(\tau)$ (KW), and is given by

$$Q_{SL}(\tau) = \sum_{j=0}^{\infty} W_z(j) \cdot \Delta t_r(\tau - j) + K \cdot \Delta t_r(\tau) \tag{7}$$

where $W_z(j)$ is heat removal weight caused by the temperature variation (temperature difference against set value) (KW/°C), it has been known and means heat removed at $\tau = jT$ when temperature indoor above set point value 1°C at $\tau=0$, K is correction factor for supply air load caused by temperature variation Δt_r, as is given by Ken-ichi Kimura[4].

$$K = -C_P \rho_r G \tag{8}$$

2.4 The Sensible Heat Output

The sensible heat output is mostly taken off by exhaust air and return air in air-conditioned room, as is given by

$$Q_E(\tau) = \rho_r C_P G t_r(\tau) \tag{9}$$

where Q_E denotes the sensible taken off by exhaust air and return air (KW).

Let ignore the temperature affection on air density, that is $\rho_O = \rho_r$. We can get the following equation by substituting Equation (5)~(9) into Equation (5)

$$t_O(\tau) = \frac{1}{TG}\left\{[V_r - \frac{T}{\rho_r C_P} W_z(0) + 2TG]\Delta t_r(\tau) - \frac{T}{\rho_r C_P} Q_L(\tau)\right.$$

$$\left. - \frac{T}{\rho_r C_P} \sum_{j=1}^{\infty} W_z(j) \Delta t_r(\tau - j) - [V_r + TG]\Delta t_r(\tau - 1)\right\} \tag{10}$$

Therefore it can be seen that supply air temperature is relevant not only to temperature variation, room volume, supply air amount, sensible cooling load at current time, but also to heat removal characteristic of system and sampling period, as well as. The room temperature variation between start-up and the time unit preceding present of the air conditioning system. To simplify calculation, it is supposed that temperature indoor absolutely satisfied the requirement of control precision, that is

$$\Delta t_r(\tau - j) \approx 0 \qquad j=1,2,\ldots,\infty \tag{11}$$

and Equation (10) can be simplified to

$$t_O(\tau) = \frac{1}{TG}\left\{ [V_r - \frac{T}{\rho_r C_P} W_z(0) + 2TG]\Delta t_r(\tau) - \frac{T}{\rho_r C_P} Q_L(\tau) \right\} \tag{12}$$

Thus the temperature variation $\Delta t_r(\tau)$ which meeting the control precision can be calculated by Equation (3), further the desired supply air temperature $t_O(\tau)$ can be obtained by Equation (12).

3. MATHEMATICAL MODEL FOR HUMIDITY CONTENT OF SUPPLY AIR

The humidity content variation indoor is described by latent heat. Supposing an air-conditioned room without free water surface, so we can get the following relationship from the law of conservation of mass:

Latent heat increment= latent heat input + latent heat produced by room - latent heat output (13)

3.1 Latent Heat Increment

$$\frac{dQ_q(\tau)}{d\tau} = \rho_r C_P V_r \frac{d_r(\tau) - d_r(\tau - 1)}{T} \tag{14}$$

where Q_q denotes latent heat indoor (KW); d_r denotes humidity content indoor (Kg/Kg$_{\text{dry air}}$).

3.2 Latent Heat Input

$$Q_{Oq}(\tau) = \rho_r G d_O(\tau) r \tag{15}$$

where Q_{Oq} denotes latent heat carried in by air supplied (KW); d_r denotes humidity content carried in by air supplied (Kg/Kg$_{\text{dry air}}$); r denotes water latent heat of vaporization (Kg/Kg).

3.3 Latent Heat Produced in Room

The latent heat within a room is mainly latent heat load in room, noted as $Q_{Lq}(\tau)$ (KW), which is a known value [3].

3.4 Latent Heat Output

$$Q_{\mathrm{Eq}}(\tau)=\rho_{\mathrm{r}}Gd_{\mathrm{r}}(\tau)r \quad (16)$$

where Q_{Eq} denotes the latent heat taken off by exhaust air and return air (KW).

Let the temperature affection on air density be ignored, Equation (17) can be got by substituting Equation(14)~(16) into Equation (13)

$$d_{\mathrm{O}}(\tau)=\frac{1}{TG}\{\ [\Delta d_{\mathrm{r}}(\tau)-\Delta d_{\mathrm{r}}(\tau-1)]V_{\mathrm{r}}+TG[\Delta d_{\mathrm{r}}(\tau)-d_{\mathrm{rg}}] -\frac{T}{\rho_{\mathrm{r}}r}Q_{\mathrm{Lq}}(\tau)\ \} \quad (17)$$

where $\Delta d_{\mathrm{r}}(\tau)$ denotes humidity content variation indoor at τ (Kg/Kg$_{\text{dry air}}$), as is given in Equation (18), d_{rg} denotes set point value for humidity content (Kg/Kg$_{\text{dry air}}$).

$$\Delta d_{\mathrm{r}}(\tau)=d_{\mathrm{r}}(\tau)-d_{\mathrm{rg}} \quad (18)$$

Similarly, for simplifying calculation, it is supposed $\Delta d_{\mathrm{r}}(\tau\text{-}1)\approx 0$, then Equation (17) becomes

$$d_{\mathrm{O}}(\tau)=\frac{1}{TG}\{\ [V_{\mathrm{r}}+TG]\Delta d_{\mathrm{r}}(\tau)-TGd_{\mathrm{rg}}]-\frac{T}{\rho_{\mathrm{r}}r}Q_{\mathrm{Lq}}(\tau)\ \} \quad (19)$$

Thus the relative humidity variation $\Delta t_{\mathrm{r}}(\tau)$ meeting the control precision can be got by Equation (3), then by Equation (20) corresponding $\Delta d_{\mathrm{r}}(\tau)$ can be solved,, furthermore, the desired supply air humidity content $t_{\mathrm{O}}(\tau)$ can be obtained by Equation (19).

$$\Delta d_{\mathrm{r}}(\tau)=0.622\frac{[\Delta\varphi_{\mathrm{r}}(\tau)+\varphi_{\mathrm{rg}}(\tau)]P_{\mathrm{q.b}}}{B_a-[\Delta\varphi_{\mathrm{r}}(\tau)+\varphi_{\mathrm{rg}}(\tau)]P_{\mathrm{q.b}}}-d_{\mathrm{rg}} \quad (20)$$

where $P_{\mathrm{q.b}}$ denotes saturated vapour pressure of moist air (Pa); B_{a} denotes atmospheric pressure (Pa).

4. MATHEMATICAL MODEL FOR ENTHALPY OF SUPPLY AIR

Enthalpy of supply air can be obtained by substituting Equation (12) and Equation (19) in Equation (19)

$$h_{\mathrm{O}}(\tau)=1.01t_{\mathrm{O}}(\tau)+d_{\mathrm{O}}(\tau)[2500+1.84t_{\mathrm{O}}(\tau)] \quad (21)$$

where $h_{\mathrm{O}}(\tau)$ denotes enthalpy of supply air (KJ/Kg$_{\text{dry air}}$).

Thus we can get enthalpy of supply air $h_{\mathrm{O}}(\tau)$ meeting the control precision of temperature and humidity indoor.

5. DETERMINATION OF FACE AREA FOR MINIMUM HEAT-EXCHANGING UNIT

Supply air enthalpy $h_{\mathrm{O}}(\tau)$ meeting the control precision of temperature and humidity indoor is the result of mixing cooled & dehumidified air and air by-pass after valve action of the minimum heat-exchanging unit. Consider the fan mixing section in Fig.2, there are

$$\rho_{\mathrm{H}}G_{\mathrm{H}}h_{\mathrm{H}}(\tau)+\rho_{\mathrm{L}}G_{\mathrm{L}}h_{\mathrm{L}}(\tau)=\rho_{\mathrm{O}}Gh_{\mathrm{O}}(\tau) \quad (22)$$

$$G=G_{\mathrm{H}}+G_{\mathrm{L}} \quad (23)$$

where ρ_{H} and ρ_{L} denote air density of mixing point H and apparatus dew point L respectively (Kg/m^3); G_{H} and G_{L} denote air volume by-pass and cooled & dehumidified air volume respectively (m^3/s); h_{H} and h_{L} denote air enthalpy of mixing point H and apparatus dew point L respectively (KJ/kg$_{\text{dry air}}$).

Let the air density variation be ignored, the following equation can be got from Equation (22) and Equation (23)

$$G_{\mathrm{H}}=\frac{h_{\mathrm{O}}(\tau)-h_{\mathrm{L}}(\tau)}{h_{\mathrm{H}}(\tau)-h_{\mathrm{L}}(\tau)}G \quad (24)$$

Besides

$$\begin{cases} G_{\mathrm{H}}=V_{\mathrm{y}}F_{\mathrm{yS}} \\ G=V_{\mathrm{y}}F_{\mathrm{y}} \end{cases} \quad (25)$$

where V_{y} denotes face air velocity of coil (m/s); F_{yS} and F_{y} denote face areas of the minimum heat-exchanging unit and the whole coil respectively (m^2). So there is

$$F_{\mathrm{ys}}=\frac{h_{\mathrm{O}}(\tau)-h_{\mathrm{L}}(\tau)}{h_{\mathrm{H}}(\tau)-h_{\mathrm{L}}(\tau)}F_{\mathrm{y}} \quad (26)$$

in which, h_{O} can be obtained by Equation (21), h_{L} is apparatus dew point under design condition, F_{y} is known value. The only unknown is h_{H}. Let's calculate h_{H} in the following way.

Consider the mixing air section of fresh and return air, there are

$$\rho_{\mathrm{N}} G_{\mathrm{N}} h_{\mathrm{N}}(\tau) + \rho_{\mathrm{W}} G_{\mathrm{W}} h_{\mathrm{W}}(\tau) = \rho_{\mathrm{H}} G h_{\mathrm{H}}(\tau) \quad (27)$$

$$G = G_{\mathrm{N}} + G_{\mathrm{W}} \quad (28)$$

where ρ_{W} denotes fresh air density (Kg/m^3); G_{N} and G_{W} denote volumes of return air and fresh air respectively (m^3/s); h_{N} and h_{W} denote air enthalpy indoor and outdoor respectively (KJ/kg$_{\text{dry air}}$).

Suppose the fresh air fraction is m, then

$$m = \frac{G_{\mathrm{W}}}{G} \quad (29)$$

Let the air density variation be ignored, the following equation can be got from Equation (27)~(29)

$$h_{\mathrm{H}}(\tau) = (1-m) h_{\mathrm{N}}(\tau) + m h_{\mathrm{W}}(\tau) \quad (30)$$

Thus the face area of minimum heat-exchanging unit F_{yS} can be figured out by substituting Equation (30) in Equation (26).

6. DETERMINATION OF THE MINIMUM HEAT-EXCHANGING UNIT

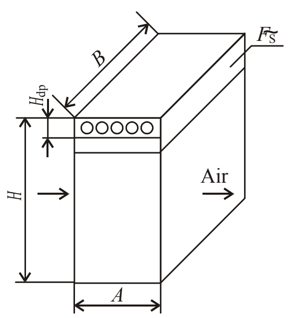

Fig.3 The geometrical structure of coil with adjustable structure

For calculating the heat-transfer area F_{S}, the geometrical structure of coil with adjustable structure should be studied. as the sketch is given in Fig.3, where A, B and H is length, width, and height respectively (m). Suppose F_{dp} is the heat-transfer area of a single row in air direction, H_{dp} is the height of a single row. Substantially, calculating F_{S}, means to find the number of F_{dp} including in F_{S}, thus

$$n_S = \frac{F_S}{F_{\mathrm{dp}}} \quad (31)$$

$$F_{\mathrm{yS}} = n_S H_{\mathrm{dp}} B \quad (32)$$

$$n_S = \frac{F_{\mathrm{yS}}}{B \cdot H_{\mathrm{dp}}} \quad (33)$$

The following equation can be got by putting Equation (33) in Equation (31)

$$F_S = \frac{F_{\mathrm{yS}}}{B \cdot H_{\mathrm{dp}}} F_{\mathrm{dp}} \quad (34)$$

Thus the heat-transfer area of heat-exchanging unit F_{S} meeting control precision can be obtained by Equation (34). In fact, Equation (33) is more useful since the face height H_{s} can be got as soon as single row height H_{dp} and n_{S} are known, as is given in Equation (35).

$$H_S = H_{\mathrm{dp}} \cdot n_S \quad (35)$$

And the face height H_{M} and H_{L} of heat-transfer area $\widetilde{F}_{\mathrm{M}}$ and $\widetilde{F}_{\mathrm{B}}$ can be got by Equation (2), as is given by Equation (36).

$$\begin{cases} H_M = 2H_S \\ H_B = 2H_M + H_S \end{cases} \quad (36)$$

So the whole coil can be divided according to unequal form as "H_{S}, H_{M}, H_{M} and H_{B}".

7. DESIGN OF THE COIL WITH ADJUSTABLE UNEQUAL STRUCTURE

According to the method of determination of the minimum heat-exchanging unit mentioned above, the design of coil with adjustable unequal structure is related to building dynamic load and design load, building volume, meteorological condition outdoor, operation of air-conditioning system, and geometrical characteristic of coil etc. The basic design steps are as follows:

(1) Calculate dynamic air-conditioning load of building, make air-conditioning project and figure out design load.

(2) Decided the type of coil with adjustable structure, including heat-transfer area, geometric size, single row area and height, and so on.

(3) Confirm the indoor temperature and humidity control precision and figure out supply air parameters meeting them.

(4) Calculate face area and face height of

minimum heat-transfer unit.

(5) Decided the segmentation of coil with adjustable unequal structure.

(6) Select double-position regulating valve for each heat-transfer unit and complete the design of the whole coil.

The steps above can be realized by developing a special software.

8. CONCLUSIONS

The variation of outdoor/indoor air-conditioning load will result in temperature and humidity fluctuating around their set values, that is, produce Δt_r and $\Delta\varphi_r$,. The purpose of air-conditioning is to eliminate Δt_r and $\Delta\varphi_r$ indoor, and make temperature and humidity meet with the demands of control precision. In general view of those factors such as temperature and humidity variation indoor, air-conditioning characteristic, building volume, meteorological condition outdoor, air-conditioning operation mode, and etc, this paper put forward the unequal structure of the coil with adjustable structure and its design method. It is not only suitable to the fuzzy control, but also satisfied with the requirements of control precision for temperature and humidity. This method seems complicated since it should consider air-conditioning characteristic as well as many other factors such as operation mode etc. In fact, an air-conditioning design with high control precision and better energy efficiency is not simply an addition, but a systematic design process based on the dynamic characteristics of air-conditioning system and building thermal system, namely dynamic design process of air-conditioning system. Only by this way, the union of building thermal system, air-conditioning system and control system, and air-conditioning running under high precision and energy efficiency can be realized. At present, coil and air-conditioning units with adjustable unequal structure have been produced by us.

The following conclusions can be got through the analyses of the paper:

(1) The structure of coil with adjustable unequal structure is reasonable and accords with the practical control action, its fuzzy segmentation are suitable to the fuzzy control and the machine-electronics realization of relevant air-conditioning devices.

(2) The minimum heat-transfer unit of coil with adjustable structure which meets the control precision of temperature and humidity in air-conditioning room can satisfy the demands of air-conditioning system control, its design idea represents the union of building thermal system, air-conditioning system and control system.

(3) The development of designing method of minimum heat-exchanging unit based on air-conditioning characteristics provides a theoretic foundation for the design of such coil and a reasonable basis for designing its software in theory.

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